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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

Relation Between Fluid Friction and Transmission Efficiency

Superheated Steam Conditions

Notes from Engineers' Handbooks

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LUBRICATION

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Relation Between Fluid Friction and Transmission Efficiency

Some of the research work carried on by The Texas Company while making a study of automobile lubrication, has been presented recently in a paper* before the Society of Automotive Engineers. This paper is given here substantially as presented except for the omission of technical details of the apparatus and test routine. Readers who would be interested in this information are referred to the original paper.

ALTHOUGH means of minimizing thermal losses in motor-vehicle engines have received a certain amount of study, very little attention has been given to the mechanical-friction losses of the engine and the transmission system. Mechanical friction is usually thought of as a necessary evil, and has been dismissed without much further analysis. An examination of the literature on tests of automobile-friction losses shows, by a lack of certain data, that all the variables were not thoroughly appreciated. As data given later in this paper will show, the quantity and viscosity of the lubricants used have a pronounced effect on the magnitude of the losses; yet most experimenters do not take this into consideration, and few even mention data from which oil viscosities can be computed. In fact, most experiments are made at either full load or full speed, apparently unmindful of the fact that a motor vehicle operates only occasionally under such conditions.

The term "mechanical losses" as used here includes all friction losses between the working

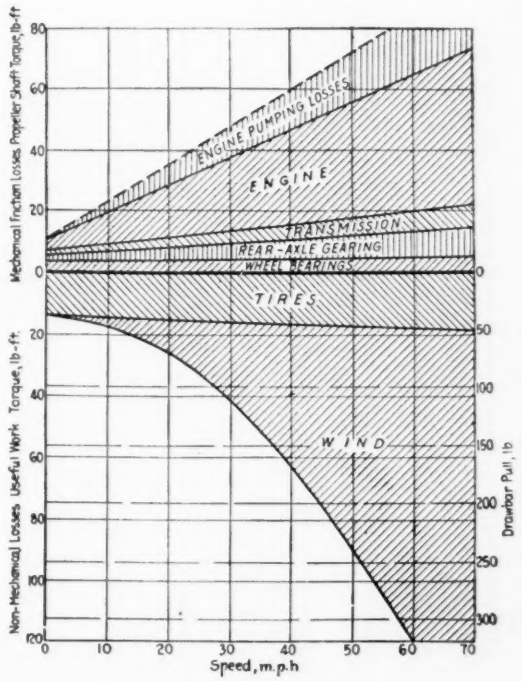

Fig. 1.—Diagram Showing the Mechanical Losses in a Touring Car Weighing 4500 Lb. and the Magnitude of the "Useful Force" Necessary to Propel the Car at Various Speeds

*By Neil MacCoull, *Automotive Engineer.*

gases in the engine cylinders and the driving-wheels of the vehicle. The only "useful work" done by an engine and its transmission system on a level road is in overcoming the resistance of tires and wind-pressure. In reality, the tire resistance is a part of the complete transmission system, but it is not a mechanical part of that system. Therefore, a mechanical efficiency of 100 per cent would mean that the indicated power of the engine would be equal to the sum of the tire and wind-pressure resistances only. The magnitude of the useful force necessary to propel a typical touring car of 4500-lb. weight at various speeds on smooth level ground, in terms of drawbar pull or towing resistance, is shown below the datum line of Fig. 1. Above the datum line, the mechanical losses of the separate units of the car are shown in terms of propeller-shaft torque. The data are computed from experiments of Prof. E. H. Lockwood at Yale, and the experiments described later in this paper. Cord tires inflated to a pressure of 65 lb. per sq. in. were assumed. Fabric tires under the same conditions would have nearly double the resistance, and both would have considerably greater resistance with lower air-pressure. The magnitude of losses due to the springs when on rough roads is not shown because no suitable data were at hand. They, like universal joint losses, are probably too small to be shown in this figure. The part of the total engine friction losses due to air friction and known as "pumping" losses has been estimated.

An examination of the items comprising the losses shown in Fig. 1 suggests four distinct possibilities for increasing the fuel economy of a motor vehicle.

(1) A suitable relation between gear-ratios and engine will reduce the well-known engine losses due to extensive throttling at average car speeds
(2) Reduction of vehicle weight
(3) Reduction of mechanical-friction losses
(4) Reduction of wind resistance

In Fig. 2, the brake horsepower, calculated from Fig. 1, to propel the car at various speeds on a level road, which checks well with actual tests, is shown, as well as the gasoline mileage at various car speeds, which has been determined experimentally for a similar car under ideal conditions on a speedway. At the lower car speeds, much greater fuel-economy would

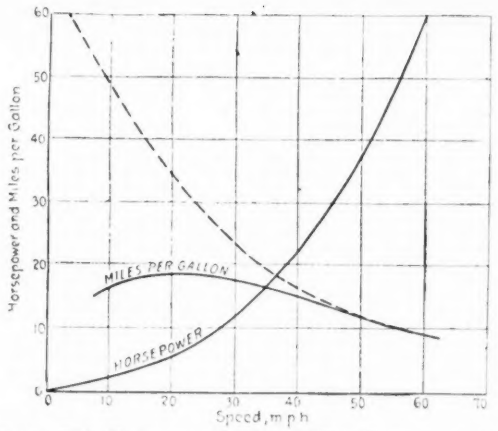

Fig. 2.—Brake Horsepower and Gasoline-Mileage Curves for a 4500 Lb. Touring Car at Various Speeds

be realized except for the very low mechanical and indicated thermal efficiencies of an engine when throttled to develop but a fraction of its normal torque. As a matter of interest, the dotted curve is given to indicate the fuel economy that could be realized if the fuel consumption per brake horsepower developed were the same at fractional as at full load. Similar results could be obtained if the engine power were no greater than just sufficient to drive the vehicle at the speed at which the economy is indicated. For instance, if the throttle had to be opened wide to make 25 m.p.h. on a level road, either because of a small rear-axle reduction or a small-sized engine, it would be possible to secure 28 miles per gallon of fuel at that speed, and better at lower speeds.

Fig. 3 shows some of Professor Lockwood's data in regard to the rolling resistance, exclud-

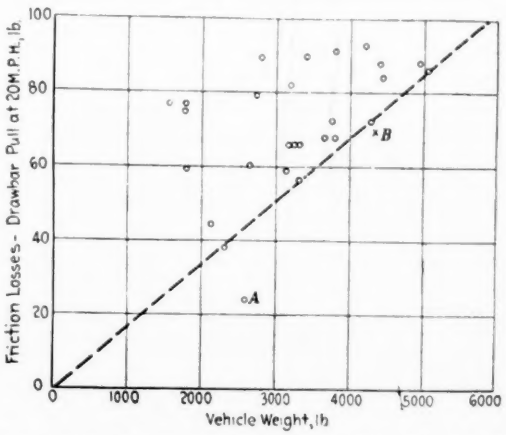

Fig. 3.—Rolling Resistance of Various Cars as They Were Actually Run Over the Road

ing wind resistance, at 20 m.p.h., of various cars taken from the road in the actual condition in which they were run by their owners. The variation of resistance with weight is evident, as well as the superiority of some cars over others of the same weight. Notice the exceptionally low resistance of car A, which made 82 miles on 1 gal. of fuel on an official road test. Point *B* represents the car assumed in Figs. 1 and 2.

The question upon which it is attempted to focus attention here is: Why have some vehicles so much less frictional resistance than others of

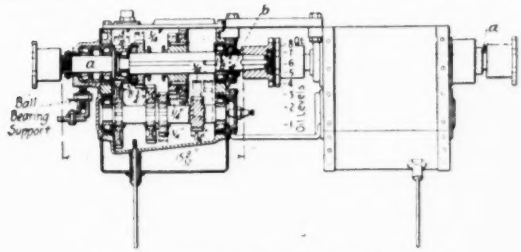

Fig. 4.—Construction of the Two Gearsets Tested

the same weight? If the distribution and cause of all these losses were known, it seems reasonable that all cars might be made with no greater resistance than those that fall on the dotted line, and even better might eventually result. Such data are food for much thought among those who wish to produce vehicles with really high fuel economy. An analysis of these losses can be made only by isolating the individual units and examining each separately.

Gearset Experiments

Methods taken to secure and develop this information for a four-speed gearset, and some

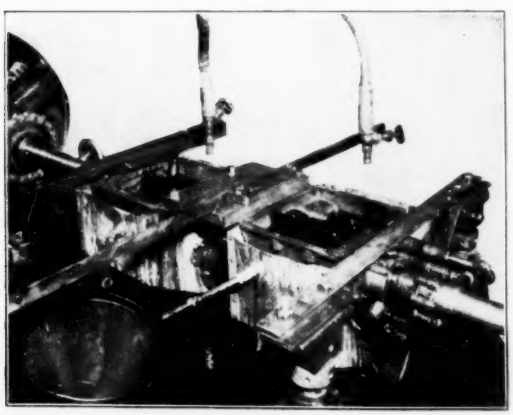

Fig. 5—The Two Gearsets as Arranged for Testing

of the results secured, are described in this paper. The data are not at all complete because of lack of time, but it is believed that the general trend of the results, even though unchecked, will be of interest to the profession. The magnitude of the task is indicated by the fact that the six variables were dealt with: gear-ratio, revolutions per minute, oil type, oil temperature, oil level, and load. No satisfactory method was available for separating the bearing losses from the gear losses and, to do so, separate bearing tests must be made. This investigation did not cover the results of

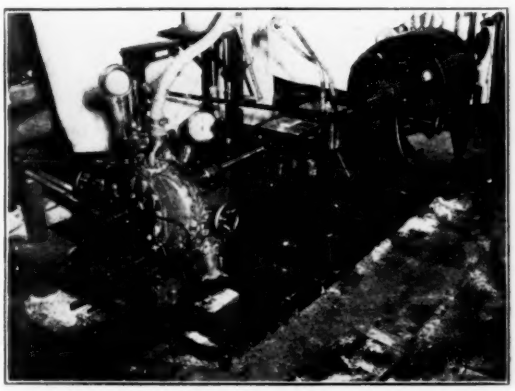

Fig. 6.—General View of the Apparatus Employed in the Tests

varying the gear-pitch, tooth-shape and tooth hardness, or the very important feature of wear.

The two gearsets tested were standard four-speed frame-type Brown-Lipe truck transmissions, rated for 35 hp. at 1000 r.p.m. The construction and principal dimensions can be seen in Fig. 4 and the gear-ratios in Table 1. The gears are of 5 per cent nickel steel, case-hardened and of 6/8 pitch. Timken taper roller-bearings were used throughout. A sheet-metal case, for water or brine and ice mixtures to control the temperature, was placed around each gear case. Thermometers were located as shown, to measure the temperatures of the oils near the center of their masses.

TABLE 1—GEAR-RATIO OF THE GEARSET TESTED

Speed	Teeth of Engaging Gears	Total Reduction
Reverse	18-33—16-13—16-35	4.81
First	18-33—16-35	4.00
Second	18-33—21-30	2.62
Third	18-33—28-23	1.50
Fourth	....	1.00

Apparatus Layout

For the sake of greater accuracy, an unusual apparatus set-up was used, by means of which the friction losses were measured directly instead of being calculated as the difference between power input and output.

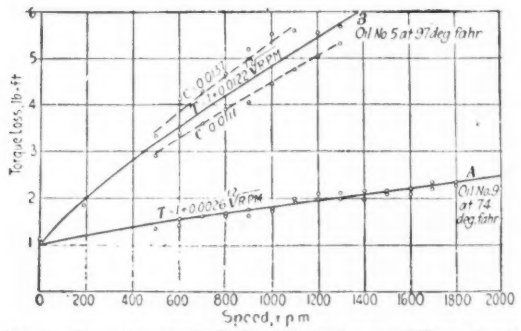

Fig. 7.—Curves Showing the No-load Friction Losses at Different Speeds

Test Results

Except where specified, all runs were taken at 1000 r.p.m., second gear and with 1 gal. of oil in each gear-case. By taking readings of losses at different speeds while accelerating and again while decelerating, so that a whole series covered less than 1 min., it was possible to discount the effects of temperature changes because the oil temperature at the end of a series was but 1 or 2 deg. higher than at the

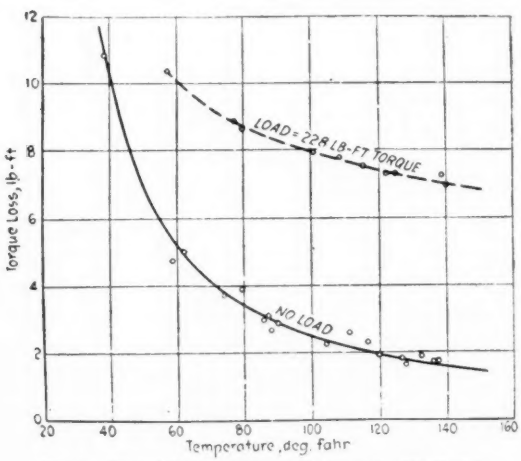

Fig. 8.—Typical Data for a Run with One of the Oils Tested at 1000 r.p.m.

start. As might be expected, the readings taken while accelerating are higher than while decelerating. Most of the runs showed a fair agreement of accelerating and decelerating data as indicated by curve *A* in Fig. 7, and an average was assumed as the true value. These curves followed an equation of the form:

$$F = K + C \sqrt[1.2]{(S)} \qquad (1)$$

where

C = a constant, depending on viscosity, oil level and gear set design.
F = friction loss in pound-feet of torque
K = a constant
S = speed of engine shaft in revolutions per minute

One of the most interesting results of these experiments was the discovery of the constant K, which is equivalent to a starting friction that must be overcome before motion begins and in addition to the losses which vary with the fluid friction of the lubricating oil. It is not certain what the cause is for the existence of this starting friction, for one does not associate

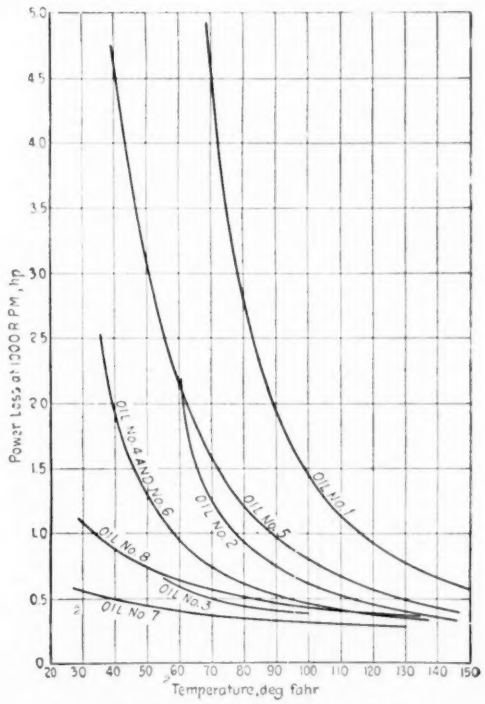

Fig. 9.—A Composite Curve of the Corresponding No-Load Losses at 1000 r.p.m. for Different Lubricants Expressed in Terms of Horsepower.

roller bearings with this characteristic of plain bearings. There is a possibility that it is due to the compression packing on the main shafts, shown at *b* in Fig. 4, that is used to prevent oil leakage, although this was not investigated. The constant C varies with the viscosity of the lubricating oil as will now be shown.

To determine the effect of temperature changes, the lubricant was chilled by packing

the jackets with crushed ice and calcium chloride.

The time required was much longer than was expected, especially when the temperature of the lubricant neared the pour-point. In one instance, a temperature of 20 deg. Fahr. below zero was maintained for nearly 3 hrs., yet the oil temperature was lowered only to 29 deg. Fahr.

The first readings were taken as soon as the motor speed was brought up to 1000 r.p.m. The motor was then stopped while the oil temperatures were read. The average reading of the thermometers in each gearcase was used in plotting the results. At the lower temperatures, the oil temperatures rose so rapidly at this speed that it was difficult to take readings with fair accuracy. Readings were taken in this way at intervals as the oil temperature rose. After a few such readings, the rise of temperature was hastened by adding warm and then hot water to the jackets. Fig. 8

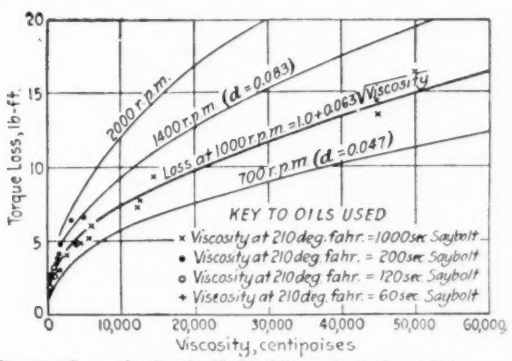
Torque Loss, lb-ft.
Viscosity, centipoises
2000 r.p.m.
1400 r.p.m. (d = 0.083)
Loss at 1000 r.p.m. = 1.0 + 0.063 √Viscosity
700 r.p.m. (d = 0.047)
KEY TO OILS USED
× Viscosity at 210 deg. fahr. = 1000 sec. Saybolt
• Viscosity at 210 deg. fahr. = 200 sec. Saybolt
o Viscosity at 210 deg. fahr. = 120 sec. Saybolt
+ Viscosity at 210 deg. fahr. = 60 sec. Saybolt

Fig. 10.—Curve Showing the No-Load Losses for Different Lubricants Plotted in Terms of Viscosity

shows typical data from such runs for one oil, and Fig. 9 is a composite of the corresponding no-load losses found for different lubricants under the same conditions but expressed in terms of horsepower. It is evident how valueless any gear-efficiency tests may be without a record of either the oil or its temperature. Fig. 10 shows the data of Fig. 9 plotted in terms of the viscosity of the oils instead of their temperatures, which is really the information desired in such a test. The curve gives results that are useful in terms of a formula for any oil, since charts are available to show their temperature-viscosity relations.[1] The heavy

[1]See *Lubrication*, June, 1921.

curve of Fig. 10 was plotted to an equation of the form:

$$F = K + D \sqrt{(V)} \qquad (2)$$

where

D = a constant, depending on oil level and gear set design.
F = friction loss in pound-feet of torque at 1000 r.p.m.
K = a constant
V = viscosity in centipoises

The constant K seems to be the same as that appearing in equation (1), although one would expect it to vary with the number of revolutions per minute. The variation of the constant D with different oil-levels will be shown later.

The losses with the oil of 200″ viscosity are consistently higher for corresponding viscosities than those of the other oils. It is believed that

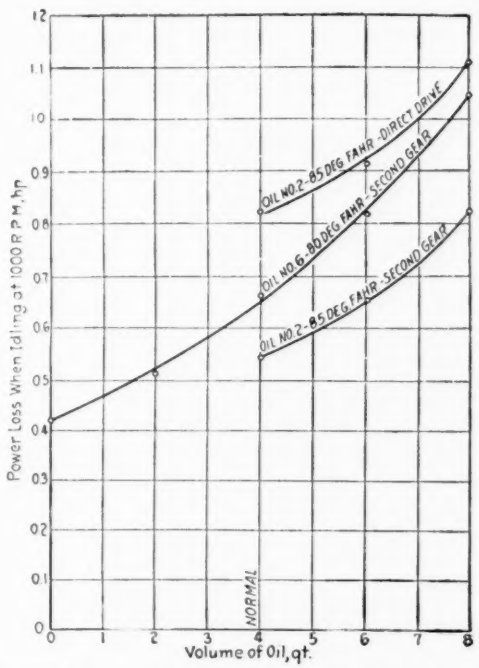
Power Loss When Idling at 1000 R.P.M., hp
Volume of Oil, qt.
OIL NO 2-85 DEG FAHR.-DIRECT DRIVE
OIL NO 6-80 DEG FAHR.-SECOND GEAR
OIL NO 2-85 DEG FAHR.-SECOND GEAR
NORMAL

Fig. 11.—Curves Showing the Effect of Oil Level on Power Losses at 1000 r.p.m.

this was due to the presence in the pocket between the engine-shaft bearings of some of the very viscous oil, No. 1, that had been tested just previously. This would be a likely source of error if the washing with kerosene were not complete. The curves shown as light lines in Fig. 10 were computed from equations (1) and (2) for a number of revolutions per minute other than the experimental readings at 1000 r.p.m. Equation (2) was used to correct much of the

following data to equivalent readings at some one temperature, since the oil temperature varied somewhat during any series of runs taking much time. Equations (1) and (2) can be combined into an equation of the form

$$F = K + [D \sqrt{V}] . [\sqrt[1.2]{S}] \qquad (3)$$

where

D = a constant affected by the oil level
F = friction loss in pound-feet of torque at 1000 r.p.m.
K = a constant
S = speed in revolutions per minute
V = viscosity in centipoises

The effect on the power losses at different levels was much less than expected. Experiments[2] made in England with a 32-hp. Leyland gearset on direct drive showed losses at 950 r.p.m. that are stated in Table 2.

Table 2—Oil Levels and Power Losses

Oil Level	Power Loss, hp.
¼	0.80
½	1.92
¾	3.20
Full	8.30

The results of our experiments were of the nature of those shown in Fig. 11 and failed to show anything like as great a variation. It was difficult to secure concordant data in this series of runs because all oils tested frothed up with air bubbles after running a few minutes so that the volume was increased materially. It was not uncommon to drain 6 qt. of oil from a gearcase that had been supplied with only 4 qt. The torque loss recorded after all oil was drained from the gearcases consisted of the constant loss K plus the viscous losses of the bearings due to the presence of a certain quantity of oil that clung to them. It is interesting to note here that the losses in direct drive are greater than in second gear for a given *engine* speed, although the reverse is true for a given *car* speed. Other experiments, such as those of Fig. 12, at a speed of 1175 r.p.m., showed that the no-load losses were lowest on first gear and increased regularly to a maximum on direct drive.

Losses Due to Load

So far, all losses given occurred while idling. While transmitting power, additional losses occur which are indicated in Fig. 13. All our experiments indicated that there is a straight-line relation between friction losses and load as shown also in Fig. 12 where similar data have been taken from experiments reported for an automobile gearset,[3] and for a steam turbine double-reduction "herringbone" gear.[4]

The intersection of the load line with the ordinate for zero load is, of course, the idling friction loss that has been analyzed and formulated during the preceding study of no-load losses. What determines the slope of this line, however, is not satisfactorily explained and is somewhat a matter of conjecture. It seems to be influenced somewhat by the viscosity of the oil, as indicated by the distance between the curves of Fig. 8. If the slope were constant, these lines would be the same distance apart throughout.

The number of revolutions per minute of the gears seems to have a similar effect, as is shown in Fig. 14, suggesting that the greater the number of revolutions is, the less the slope becomes. The gear-ratio also has its influence; the greater the ratio is, the greater the slope becomes, but this would be expected from the greater tooth-pressures and bearing loads. Light lines are shown in Fig. 13 that represent various efficiencies. The load curves cross such lines, giving the usual characteristic of an increasing efficiency with increasing load. If the load were continued indefinitely, the efficiency would approach but never wholly equal the efficiency represented by one of these constant-efficiency lines running parallel to it. For the curve representing 1000 r.p.m., this optimum efficiency would be about 98 per cent. Thus, the higher the load is, the higher the efficiency will be until a load is reached where the load curve ceases to be straight and becomes concave upward, a point not reached in these experiments.

Since the efficiency increases with load, it is interesting to know how high a load the gears can carry. When delivering 43.3 hp. at 1000 r.p.m., which represents 227 lb.-ft. of torque, the tooth load was over 4000 lb. per lineal in. of face-width and the static fiber-stress was about 67,000 lb. per sq. in. This stress was figured from a formula given by Prof. G. H. Marx

[2]See Report of National Physical Laboratory for 1913-1914, p. 101.

[3]See *Mechanical Engineering*, November, 1920, p. 613.
[4]See *Journal of the American Institute of Electrical Engineers*, September, 1921, p. 724.

as a result of his experiments.[5] Over 50 hp. was carried several times with no sign of distress, which corresponds to a tooth load of 4800 lb. per lineal in. and a fiber stress of over 80,000 lb. per sq. in. Static tests show a breaking load of 18,000 lb. on a ¾-in. face, which corresponds to a fiber stress of 305,000 lb. per sq. in. These figures are so very much higher than experienced in other branches of the engineering profession that they are especially noteworthy. It is probable that failure by abrasion or wear will occur before breakage, but there was no sign of abrasion in these tests. Experiments on the wear of gears loaded to 3668 lb. per lineal in. have been made and the results recently published.[6] In making use of these figures, it must be borne in mind that these loads were all uniform and not fluctuating as would be the case if the drive had been by a four or even a six-cylinder gasoline engine, in which case a similar maximum would result from a lower average load. Notice that the optimum efficiency will be approached at all loads by a reduction of the no-load losses. If there were no losses at no load, the efficiency would remain constant and equal to the optimum at all loads.

Efficiency of the Gearset

When referring to gearset efficiency it is only fair to talk of the efficiency at the loads experienced in operation. This has been done for level-road conditions in Fig. 14, taking the loads transmitted from Fig. 1 as the sum of the wind, tire, wheel-bearing and rear-axle losses at various speeds.

The curves for direct drive at different temperatures are of particular interest because of showing lower efficiency, with the "heavy" gear-oils now popular, than is usually thought of except when in gear. When a vehicle is run only a few miles between stops during cold weather, the oil does not have time to heat up much and may easily be at a temperature no higher than 60 deg. Fahr. The shape of these curves at first seems odd, but is explained by the fact that at low speeds the losses increase more rapidly than the load, accounting for the falling efficiency with increased car speed. The phase of rising efficiency with increasing car speed results when the load increases faster than the losses. In addition to the curve given for second-gear efficiency, a dotted curve is given for the same losses, while carrying the full rated torque of the gearset. Since gears are rarely used except with wide-open throttle when climbing hills, this represents more nearly the actual efficiency obtained than the corresponding curve for level-road conditions. A similar increase of efficiency results for direct drive when climbing grades requiring full throttle, but the level-road conditions more nearly represent average conditions.

[5]See *Transactions of the American Society of Mechanical Engineers*, vol. 37, p. 520.

[6]See *Automotive Industries*, Nov. 3, 1921, p. 865.

Noise Measurements

Noise is such an important factor in the transmission of power by gears that an attempt

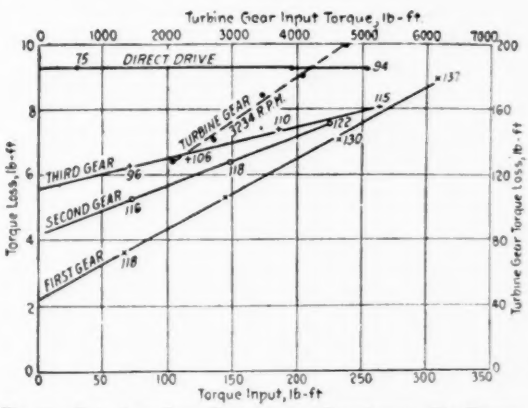

Fig. 12.—Data from Some Contemporary Experiments. The Figures Given on the Curves Are the Oil Temperatures

was made to measure it and determine how it is influenced by the lubricant used. After consulting several authorities, the following apparatus was suggested by the Western Electric Co., who were also kind enough to lend it. A microphone of the inertia type, without a diaphragm, was connected in series with a single cell of storage battery and the heating coil of a thermocouple, the thermocouple itself being connected directly to a direct-current microammeter. With the microphone button in contact with the outside of the gearcase, the microammeter pointer gave a reading that was recorded as a measure of the energy of vibration of the case, and was assumed to be a measure of the noise produced by this vibration. As slightly different readings resulted from contact

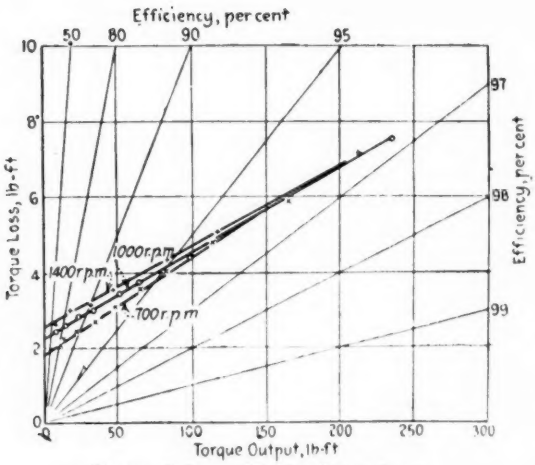

Fig. 13.—Influence of Load on Friction Losses

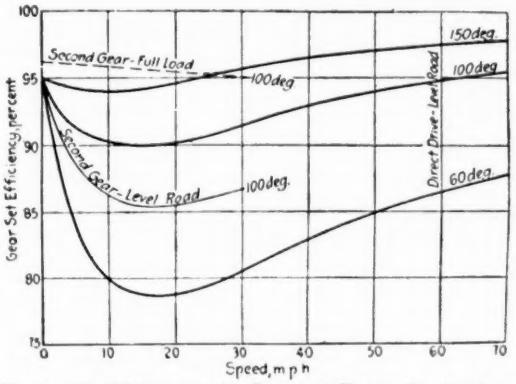

Fig. 14.—The Efficiency of the Gearset at Various Car Speeds on Direct and Second Gear. The Figures at the Ends of the Curves Indicate the Oil Temperatures

with different parts of the case, contact was made at approximately the same point, at a corner, for all subsequent readings. Fig. 15 shows the results on direct drive and also on second gear. The pronounced crest of these curves, when one would expect them to continue upward, indicates that the microphone was not adapted to periods of vibration as high as those experienced. A change in the microphone was made but this was not much of an improvement. It is clear that further development will be necessary to make this apparatus useful, and there is little doubt that this can be done.

Summary

It has been shown that friction losses in a gearset consist of two distinct types; (*a*) those existing while idling or transmitting no power and (*b*) additional losses while transmitting power. The former were shown to be the sum of a constant and a factor for the fluid friction that varies with the viscosity and quantity of oil as well as with the number of revolutions per minute of the gears. When it is desirable, these factors can be formulated for a gearset so that the losses can be calculated with reasonable precision for all combinations of the variables involved. It has been shown also that an appreciable increase in efficiency will result by lowering the viscosity of the oil, particularly when the gearset is used for direct drive on a level road. Recommendations to use oils of low viscosity cannot yet be made, however, because of insufficient knowledge of the rate of wear and volume of noise resulting from their use. It is doubtful if there will be any noticeable increase in the wear of an automobile gearset if an oil similar to a heavy motor oil be used, particularly since these gears are not used continually. The De Dion-Bouton trucks as well as the Lanchester automobile have made very successful use of such an oil for their gearsets, and of course the transmissions of Fords get only such oil as used in the engine.

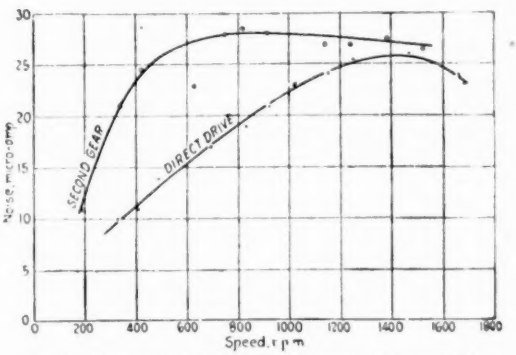

Fig. 15.—Results of the Unsuccessful Attempt to Measure Noise at Various Speeds

The data that have been given here are very suggestive for further analytical study, in regard to both gearsets and other parts of the power system of a motor vehicle. Time thus taken promises well to increase the efficiency of motor-vehicle transportation and reduce its cost, as well as to make our fuel resources last longer.

Compounded Cylinder Oil for Superheated Steam Conditions

A GREAT many engineers have encountered considerable trouble with the lubrication of steam cylinders using superheated steam, apparently because they have neglected to remember that only in a few cases of very high superheat, is there any superheat in the steam at the end of the expansion stroke, or throughout the entire exhaust stroke. It is believed that something like 200 degrees initial superheat at the throttle is required for any superheat to remain in the steam at exhaust when operating at about ¼ stroke. This means that practically all engines using superheat are operated on saturated steam for a portion of their expansion stroke and through the entire exhaust stroke. Consequently, instead of a purely superheated steam condition, we have two conditions to meet, one of superheated steam and one of saturated steam.

Most of the trouble encountered in the lubrication of superheated steam cylinders has been caused by the carbonization of the oil and deposits on the valves at the end of the valve travel and at the end of the counterbore in the cylinders. These carbon deposits are caused by excessive quantities of oil being fed to the cylinders and can be eliminated if the quantity of oil fed is reduced. The carbonization is caused by the oil remaining in contact with the hot surfaces and the superheated steam long enough for its lighter constituents to be evaporated and the oil partially decomposed. If the oil were of such characteristics that it evaporated more readily and if it were fed to the cylinders in such small quantities that an accumulation of the oil would not be made on the valves and other parts of the engine, the carbon deposits could be entirely eliminated.

The excessive oil feeds which many engineers have considered necessary for superheated steam cylinders, are due to the use of a straight mineral cylinder oil of high viscosity and high flash and fire. This straight mineral oil, i.e., a cylinder oil without compounding, while suitable for the lubrication of the cylinder, so long as the steam remains dry or superheated, is not suitable for the lubrication of the cylinder during that portion of the stroke when the steam is saturated, as the moisture washes the oil off the cylinder walls and the cylinder becomes dry. This results in wear and increased friction. Consequently the engineer increases the amount fed, so that something like satisfactory lubrication is secured, and in doing so, such a large amount of oil is fed to the cylinders that it is not carried away, and accumulations and carbon deposits result.

The proper oil for a superheated steam cylinder is a medium-heavy viscosity cylinder oil, having a fair amount of compounding, something like 4 or 5% of animal oil. Such an oil, on account of this compounding, will emulsify slightly and thereby lubricate the cylinders very efficiently during that period when they are filled with saturated steam. As the compounding is small there will be no bad effects resulting from the exposure of the oil to superheat conditions. It is fully appreciated that this is somewhat contrary to the usual understanding and the general recommendations as to the properties necessary for an oil to withstand superheated steam, but the success which has attended the use of this type of oil, makes the explanation given above appear as the proper one. Of course, there are conditions where the use of a compounded oil may not be advisable, such as in the reciprocating engines used in marine practice. However, since the majority of the cylinders of such engines are vertical, and a very small amount of oil is required for their lubrication, it is not believed that any difficulty will arise from the use of a straight mineral cylinder oil under superheat conditions. In general, it is therefore recommended that a slightly compounded oil be used in cylinders subjected to superheated steam conditions.

Lubricating One End of Steam Cylinder

SOME time ago I was interviewing the Chief Engineer of a large manufacturing plant in regard to lubrication, and he mentioned the fact that the exhaust valve on the crank end of the high pressure side of their 32 x 64 x 48″ Cross-Compound Corliss engine always made a groaning noise, regardless of the amount of oil fed.

An examination of the unit was made during the noon hour when it was shut down. When the engine was stopped, the valve cover plate was removed, and it was found that the end of the valve was worn very bright, and had worn into the cover plate about ⅛ of an inch. The valve appeared to be getting very little lubrication, although the seat did not show any signs of being worn. The cover plate was placed in a planer and machined off smooth. Two oil grooves, or steam grooves, were cut in the top of the valve, and the valve was then replaced. It was found that there was $\frac{3}{16}$ inch clearance between the cover plate and the end of the valve. Upon starting the engine it ran very quietly for the rest of the afternoon, and naturally it was thought that the trouble had been overcome.

About three weeks later I had occasion to call at the plant again, and made inquiries in regard to the noisy exhaust valve. The Chief Engineer advised that the noise had been eliminated but they were still breaking valve stems. I was very much surprised, as he had not mentioned this fact on the occasion of my previous interview, and advised that I would like very much to have an opportunity to make a thorough investigation the first time the engine was shut down. As they had lost seven valve stems on this particular valve during the past two years, and as the builder of the engine had put two men on the job without being able to solve the mystery, the Chief Engineer was glad to comply with my request. As it happened to be Saturday, and the plant would be shut down at noon and not started again until Monday morning, we both agreed that it was an ideal opportunity for a thorough investigation. A mechanic was furnished as a helper, and the bonnets from the exhaust valves on either end of the high pressure cylinder were removed, and it was found that the valve on the head end was receiving an excess amount of oil, while the one on the crank end appeared to be almost dry. As the cylinder had very large valve ports we slipped an electric light into the cylinder through the ports, and found that the crank end of the cylinder was very dry and looked as though it had received little oil. The head end of the cylinder, however, had a very nice polish on it, and there was oil clinging to the walls. With this information available, inquiries were made as to how the oil entered the steam, and whether there was anything on the inside of the steam line to aid atomization. The mechanic advised that there was a ⅜ inch pipe with the end shaped like a spoon, that extended into the center of the steam line. Naturally it was thought that this had been broken off, or become loose and worked out, but the mechanic advised that he had had it out when they had changed the oil feed from under to above the throttle. Not being satisfied that the trouble did not lay with this atomizer, I marked the bushing that held it, so that I would know just what position it was in before removing it. My surprise can well be imagined when I found that the quill or spoon as they called it was upside down, so that the oil that fed through it and out at the end was practically protected from the flow of steam. With this arrangement it can readily be seen that the oil followed the under side of the quill back to the steam pipe, where it ran down the side of the pipe to the steam chest, and as this lead down to the head end side of the opening the oil was carried to that part of the cylinder, leaving very little, if any, for the crank end.

The spoon was replaced in the steam line in the proper position, and the engine was started. During the following six months I had several opportunities to examine the cylinder, and each time found it was receiving a sufficient amount of oil, although the consumption had been reduced 50%.

The above brings out two facts: 1st: Investigation should not be made unless there is ample time to make it thorough; 2nd: Steam cylinder oil not properly fed into the steam, will not furnish efficient lubrication regardless of the quality of the oil.

Notes From Engineers' Handbooks

Do You Know Your Oil Pressure?

A short time ago one of our municipalities purchased a 200 H. P. Reduction Geared Turbine, driving a Centrifugal Pump. As we were taking care of their lubricating requirements at that time one of our engineers was called in to make recommendations as to the proper oil to use for this outfit. Taking into consideration that it was a gravity feed system with reduction gears using the same oil as the bearings, an oil of 300″ Saybolt viscosity at 100° F. was recommended.

As mentioned before the reduction gears of the turbine were lubricated entirely by gravity feed, the tank being placed approximately 16 feet above the floor level; therefore there was about a 12 foot drop to the upper bearings of the turbine. When the turbine was started for the first time considerable trouble was experienced through hot bearings, and immediately the trouble was blamed on the lubricant. The engineers of the plant claimed that the oil was too heavy, and got in touch with the manufacturers of the unit who told them an oil with a viscosity of at least 200″ Saybolt at 100° F. should be used. As the oil we had recommended was 300″ viscosity, it was evident that the trouble could not be blamed on the high viscosity of the oil.

The trouble with the supporting bearings was soon eliminated by getting them into proper alignment, but heating still persisted in the thrust bearings on the steam end of the turbine. It was difficult here to determine whether this heating was due to friction or conduction from the turbine proper, but the inclination of the engineers of the plant was still to blame it upon the oil. Our Lubrication Engineer then took it upon himself to make a complete inspection of the lubricating system. He discovered that the line from the gravity tank came down directly and entered the reduction gear case with full outlet. The supply to the thrust bearings, which were 4-5 feet to one side, led off near this outlet to the reduction gear case. When pressure gauges were installed in the line it was found that there was practically no pressure of oil at the thrust bearings, and they naturally were not receiving proper lubrication. The piping was changed so that instead of going directly to the gear case the oil went directly to the thrust bearings and then to the gears, with a proper size outlet at the bearings so as to give the correct pressure of oil both to the bearings and the gear case. From this time on no further trouble was met, and everybody was pleased with the oil.

Cold Makes Hot

A complaint was made to one of our offices that the last barrel of oil which a large manufacturer had received did not work satisfactorily, as the eccentrics on one of their large Corliss Engines using this oil had been running hot ever since they had started to use the oil from this particular barrel. One of the company engineers was sent to investigate this complaint, and found that the eccentrics certainly were hot, and for no apparent reason.

As he was standing near this piece of machinery trying to study out the cause, he could feel a draft of cold air rushing in behind him, caused by a broken window near the machine. This window was only 6 feet away from the eccentrics and they were in direct line with the draft. The temperature outside was zero. Our engineer covered the window with a piece of pasteboard, and very shortly afterward the eccentrics cooled and ran without further trouble.

This goes to show that cold air blowing on a particular portion of a machine may so cause unequal contractions in the metal parts as to throw them out of an adjustment which was made when the temperature was normal.

Needless Waste of Lubricants

We are quoting below an editorial from AUTOMOTIVE INDUSTRIES of June 30, 1921, which we believe should be noted by every car owner. The moral is self-evident.

"A mark of careless design or assembly is the pool of oil left on the garage floor or roadway by many cars even though they be new and presumably in good repair. A floor pan is required in many cases to prevent the floor of the show room from becoming stained, while our highways are black with oil dropped from cars and trucks which use them. A dollar a gallon is, one would think, a rather high price to pay for road oil, yet that is about what the average motorist pays for oil, a large proportion of which is ultimately deposited on the roadways wherever cars are used.

"This would be bad enough, simply because it represents a waste which is in large part preventable, were lubricating oils cheap and the natural supply unlimited, but when the price of good oil is high and the supply decidedly limited the waste is not to be condoned, especially when there are many ways of minimizing if not eliminating the waste; such as thrower rings and flanges, felt washers, return threads, proper gaskets, better inclosure, with return ducts for oil, etc. It is rightly considered a mark of poor design, construction or repair to see oil leaking from any machine in a well-kept shop or power plant. The mere fact that in the modern car the oil leak is hidden is no excuse for continuing a design known to be defective in this regard. This is a subject that may well receive more study on the part of the engineering, production and service departments of all producers of automotive equipment."

Settling Complaints for the Other Fellows

Recently our Engineering Force was called upon by the manager of a manufacturing plant to investigate the lubricating conditions of one of the engines under his supervision. He was using a competitor's oil and had experimented with several grades but was unable to get satisfactory lubrication as the oil seemed to vaporize to an unusual degree, settling on the equipment in the room. He wished our engineer to furnish him with an oil that had sufficiently high flash and fire points so that it would not vaporize when used in the circulating system of the Corliss Engine. Our engineer, feeling that such a course was inadvisable and knowing that there must be some contributory cause to this vaporization, made an investigation of the actual working conditions of the engine. This engine was a 22 x 42 Corliss running at a speed of 150 r. p. m. and driving a generator. It was equipped with a well-known individual filtering and circulating system which was investigated and found to be operating satisfactorily. The direction of rotation of the engine was over, and the cross-head guides were equipped with a sheet iron shield which was hinged to the bottom guide and extended up to within two inches of the top guide. This arrangement so constricted the volume within the shield that when the machine was operating there was a very high air displacement on the forward end of the guide. They were lubricating the guides by means of a ⅛ inch stream of oil through the top guide so as to lubricate the top cross-head shoe which, by the way, did not touch the guide. The quantity of oil was so large that the apparatus was practically being lubricated with the splash system, and due to the high velocity of the air passing through the two inch opening between the top guide and sheet iron shield a very fine spray of oil was carried out into the room. This air atomization was accentuated by the contact of the oil with the hot piston rod. The trouble was corrected by feeding the oil to the top of the cross-head a drop at a time, and doubling the opening between the shield and top cross-head guide so as to reduce the velocity of the air and eliminate atomization. It was found under these conditions that our competitor's oil was satisfactory, but he lost out on account of not being on the job in ascertaining the cause of the trouble.

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A SUPERHEATED DISCUSSION

Between:

HI PRESSURE, An Engineer with a triple expansion grouch AND A TEXACO MAN

H. P.: "Never mind about the weather—just you tell me why I get carbon in the ports and clearances of that there coffee pot that calls itself an engine."

Texaco: "You use too much oil—"

H. P.: "That's a bright remark. I'd like to see you cut down the oil feed—and keep the low pressure cylinders and valves from groaning and complaining like a pig under a tub."

Texaco: "That's it—you're trying to lubricate the low pressure end by shooting in a lot of oil with the steam—at the high pressure end."

H. P.: "Say, will you get this into your head—if I feed less oil I don't get ANY lubrication in the low pressure!"

Texaco: "I know that."

H. P.: "Then why—"

Texaco: "I'll tell you. But first let me ask you a few questions. Alright, then. How dry is superheated steam?"

H. P.: "Go on! How old is Ann? Do you know any more funny ones?"

Texaco: "I'm not joshing—let me go after this in my own way—I think I can show you what's what."

H. P.: "Go ahead, but cut the comedy."

Texaco: "Now then, tell me, do you get any saturated steam at the end of the high pressure expansion stroke?"

H. P.: "Mebbe so!"

Texaco: "Do you get any during the exhaust stroke?"

H. P.: "Y-E-S."

Texaco: "Do you have saturated steam in the intermediate?"

H. P.: "Sure!"

Texaco: "Do you get condensation in the low pressure and on the low pressure exhaust valves?"

H. P.: "You bet."

Texaco: "Don't saturated steam and condensation mean moisture? You agree. Alright. Now, I'm coming back to that again. But tell me what kind of oil are you using?"

H. P.: "Blame good oil and you know it."

Texaco: "Yes, but it's not the right kind."

H. P.: "Of course YOU would say that."

Texaco: "Isn't it a straight mineral oil?"

H. P.: "Yes."

Texaco: "Well, here's my point. You know that straight mineral oil will not stick to wet metal surfaces. You know it doesn't. You prove that yourself when you try to compensate for that lack of sticking quality by feeding a lot of oil—to try to get enough —"

H. P.: "I'm beginning to get you. Compounding?"

Texaco: "Wait a second—what happens?—the excess oil collects in the ports—in the counterbores—gets baked by the heat—carbonization."

H. P.: "So that's it."

Texaco: "What you need is an oil that will stick to moist metal—then you wouldn't need so much—only enough to maintain a film—and then there wouldn't be any excess oil. No excess. No carbon."

H. P.: "But a lot of engineers say "use straight mineral oil for superheated—"

Texaco: "Ha! that's it—THEORETICAL superheat—but you're up against REAL moisture, so you have to use an oil that has enough compounding to stick."

"Then you can cut down your feed and still keep a perfect film—that will show by the fine operation and entire absence of groaning in low pressure cylinders—and remember: No excessive oil—no carbon."

We'll tell the rest of it:

For superheated steam use

TEXACO LEADER CYLINDER OIL

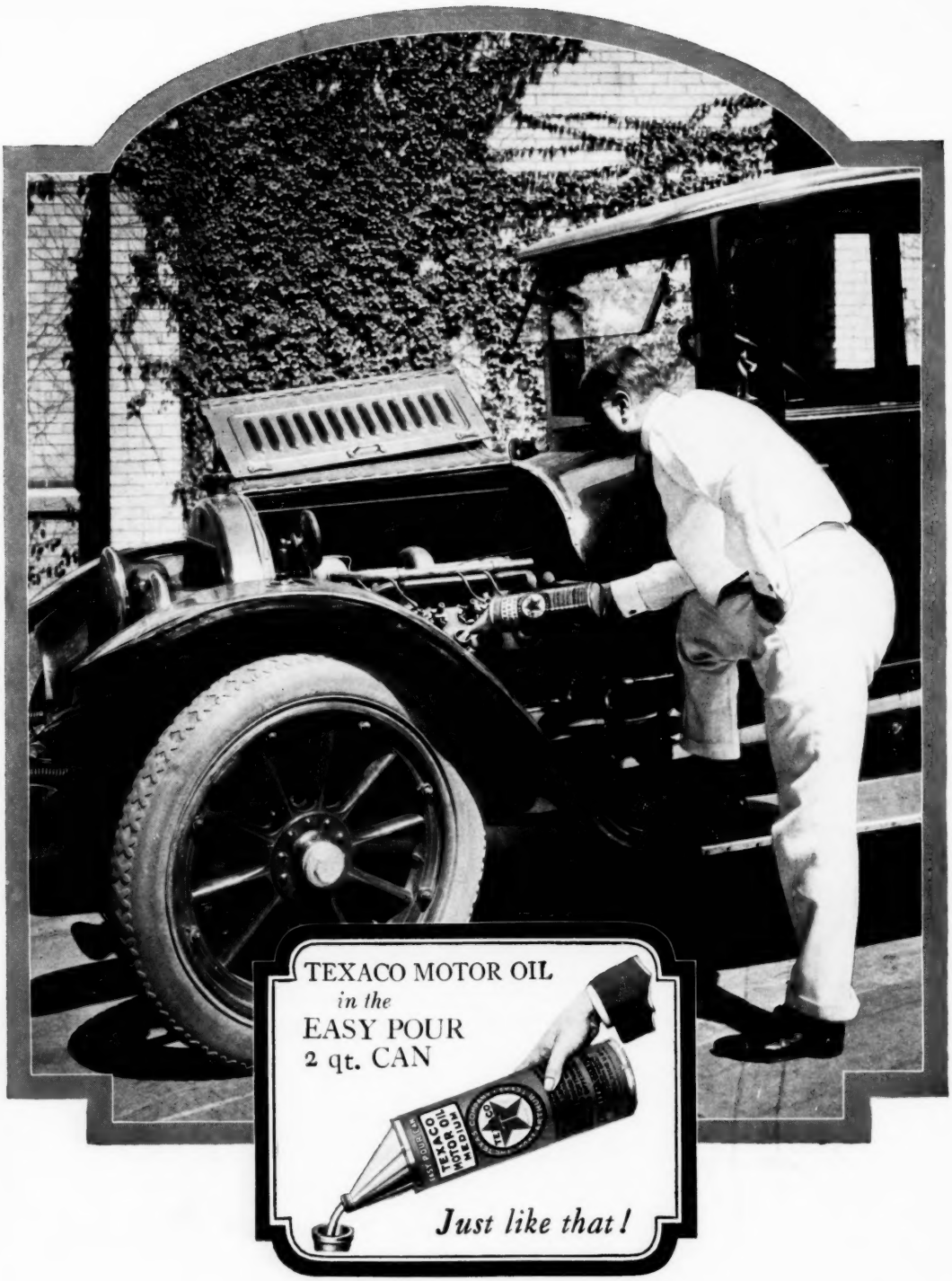
or

TEXACO 650 T CYLINDER OIL

They lubricate perfectly, atomize easily and adhere properly to the moist metal surfaces.

AND you need to feed only very little oil—that's good lubricating practice.

There is a Texaco Lubricant for Every Purpose



THE TEXAS COMPANY

Texaco Petroleum Products

Dept. H, 17 Battery Place, New York City

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